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Tozzi et al.

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(54) **TOROIDAL VORTEX INDUCTION DIFFUSER**

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(73) Assignee: **Prometheus Applied Technologies LLC**, Fort Collins, CO (US)

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(51) **Int. Cl.**

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F02M 21/04	(2006.01)
F02M 29/06	(2006.01)
F02M 35/104	(2006.01)

(52) **U.S. Cl.**
CPC ***F02M 35/10262*** (2013.01); ***F02M 21/042*** (2013.01); ***F02M 21/045*** (2013.01); ***F02M 29/06*** (2013.01); ***F02M 35/104*** (2013.01)

(58) **Field of Classification Search**
CPC F02M 35/10262; F02M 35/104; F02M 35/1211; F02M 35/1216; F02M 21/0218; F02M 21/04; F02M 21/042; F02M 21/045; F02M 29/00; F02M 29/06; F02M 2700/126
See application file for complete search history.

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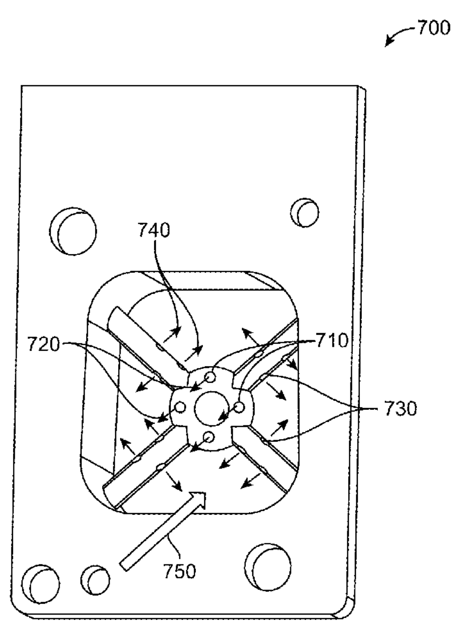
(74) *Attorney, Agent, or Firm* — Barcelo, Harrison & Walker LLP

(57) **ABSTRACT**

In certain embodiments, a PFI diffuser induction device may use toroidal vortex flow to thoroughly mix H_2 and air in the intake runner and port of a H_2 engine. When H_2 enters a stream of air flow in the form of a toroidal vortex, it may tend to swallow the air into the vortex where a low-pressure region may be formed due to the swirling velocity of the vortex, which may be more effective that the typical mixing via conventional injection methods. Engine test measurements show remarkable improvements in engine combustion stability as well as engine efficiency and power output using a counterflow Toroidal Vortex Induction Diffuser to achieve high levels of fuel mixture homogeneity in combustion engines using hard-to-mix fuels like H_2, CH_3OH, C_2H_5OH and other gaseous and liquid fuels.

19 Claims, 9 Drawing Sheets



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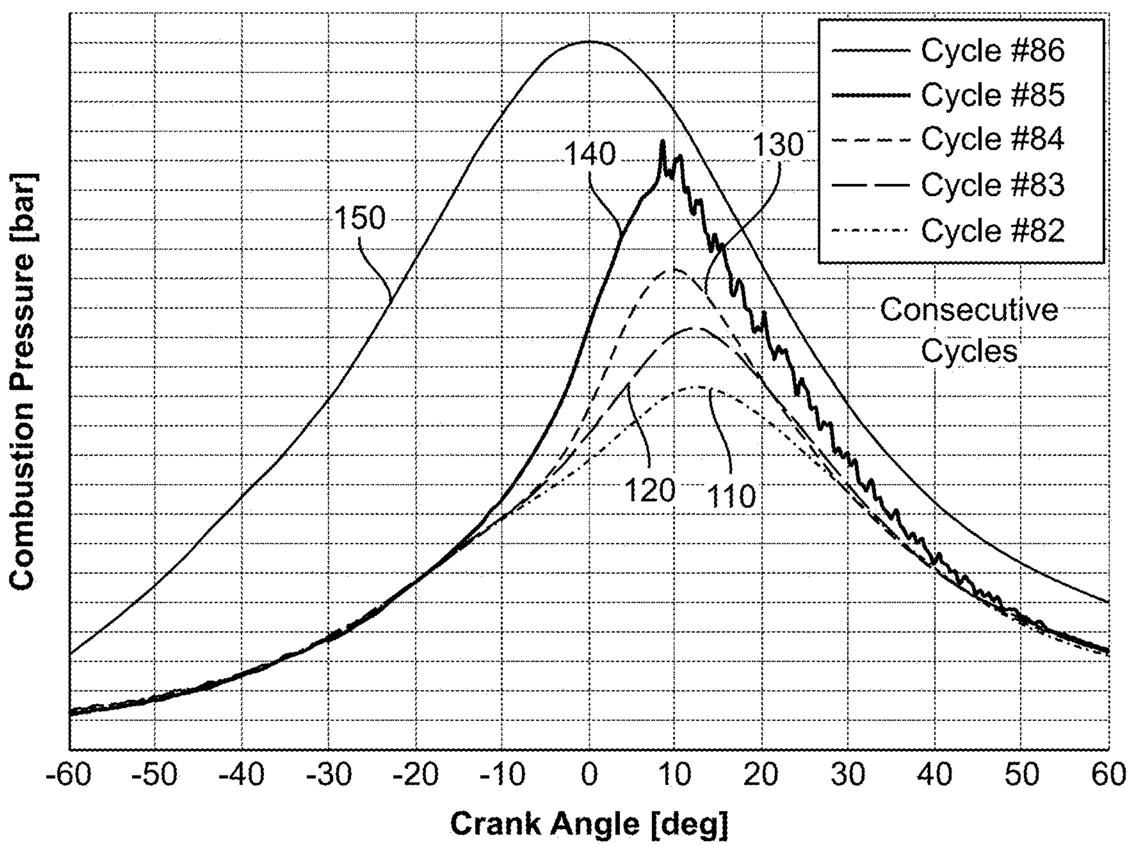
Cycle #86
Cycle #85
Cycle #84
Cycle #83
Cycle #82
150
140
130
120
110
Consecutive Cycles
Combustion Pressure [bar]
Crank Angle [deg]
-60
-50
-40
-30
-20
-10
0
10
20
30
40
50
60

FIG. 1

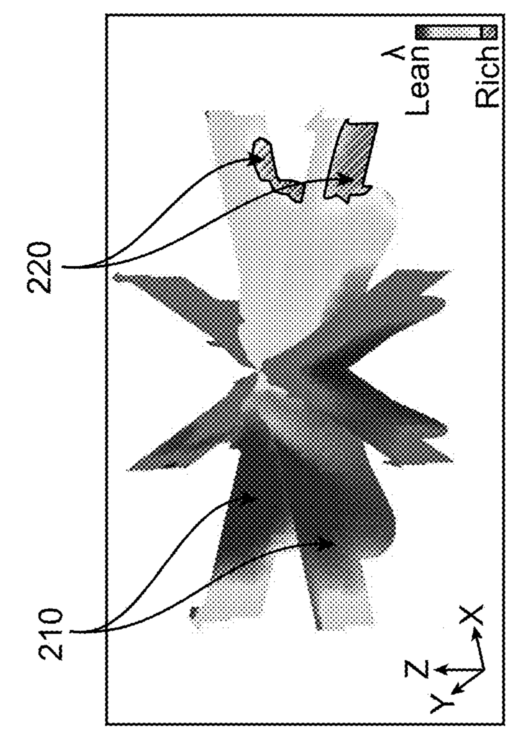

FIG. 2

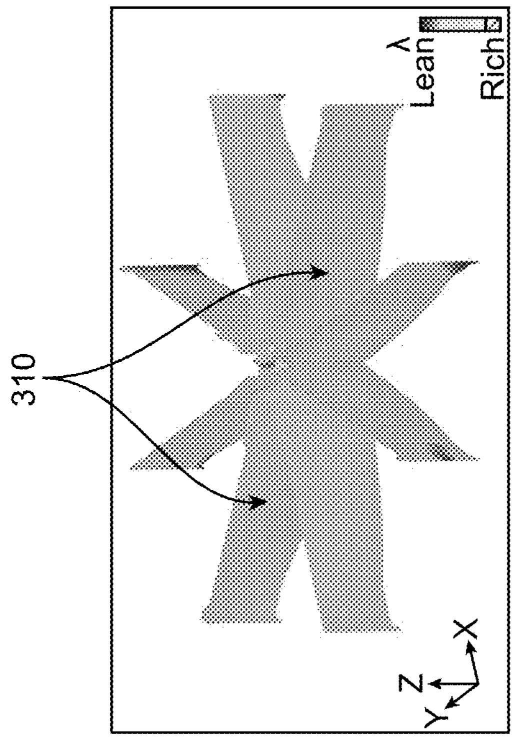

FIG. 3

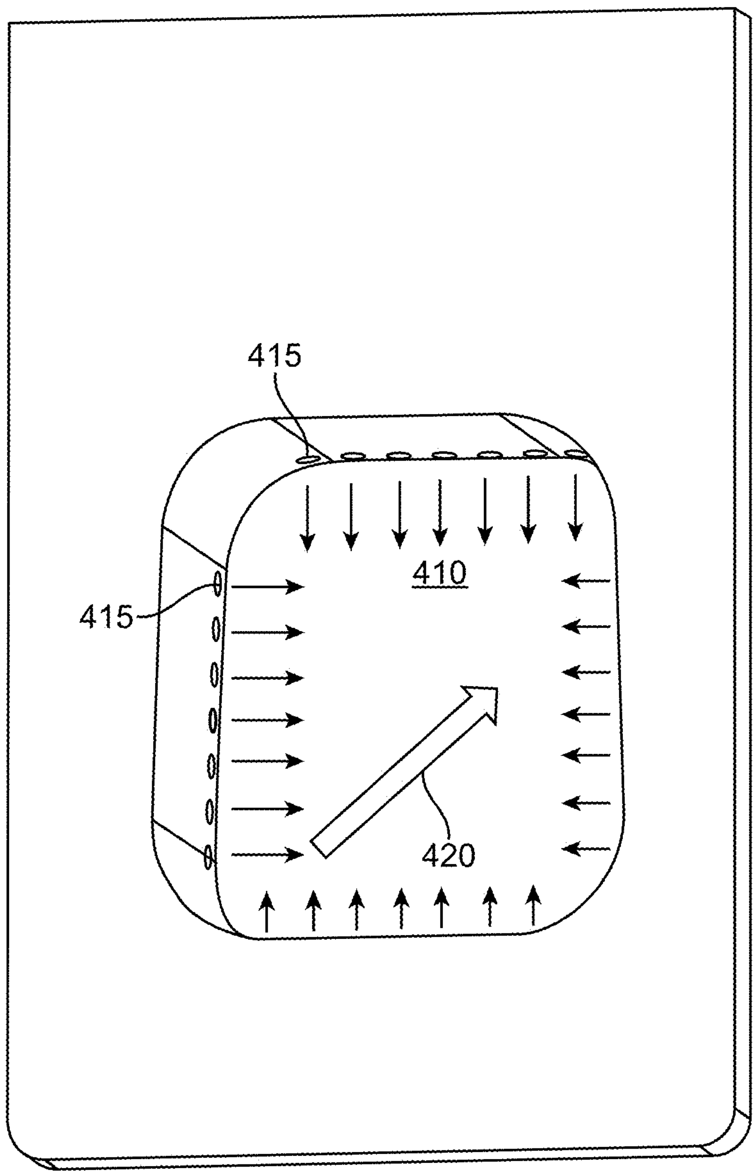

FIG. 4

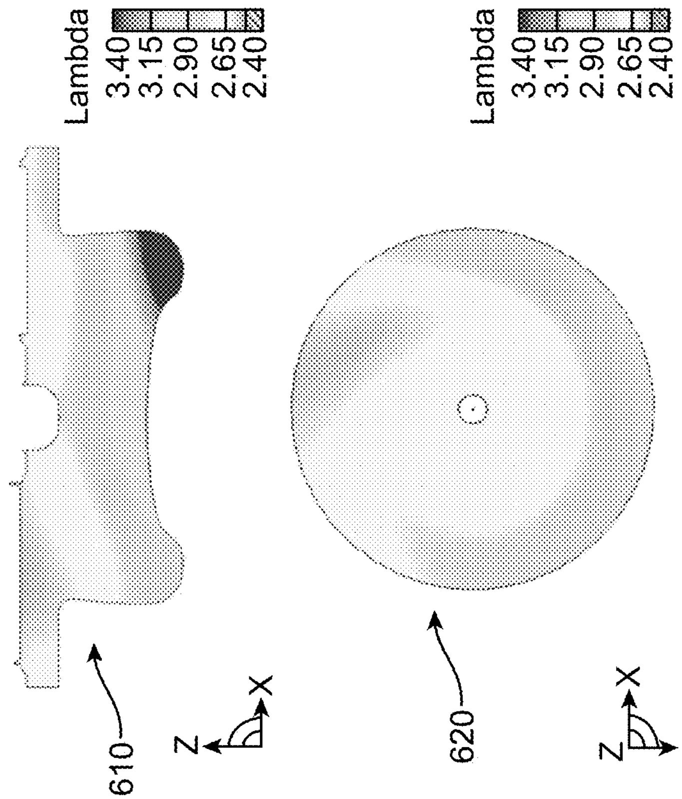
Lambda
3.40
3.15
2.90
2.65
2.40
Lambda
3.40
3.15
2.90
2.65
2.40
610
620
Z
X
Z
X

FIG. 6

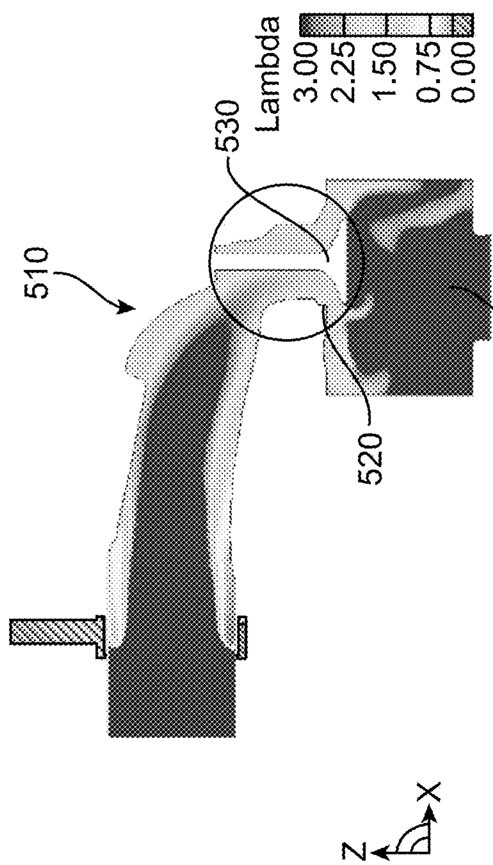
510
530
520
540
Lambda
3.00
2.25
1.50
0.75
0.00
Z
X

FIG. 5

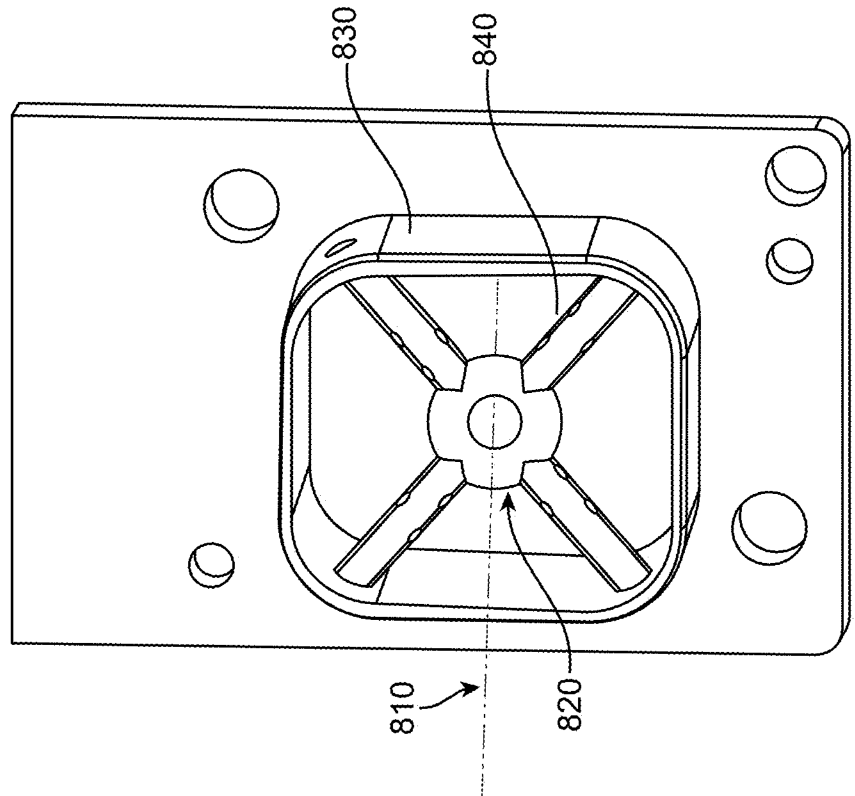

FIG. 8

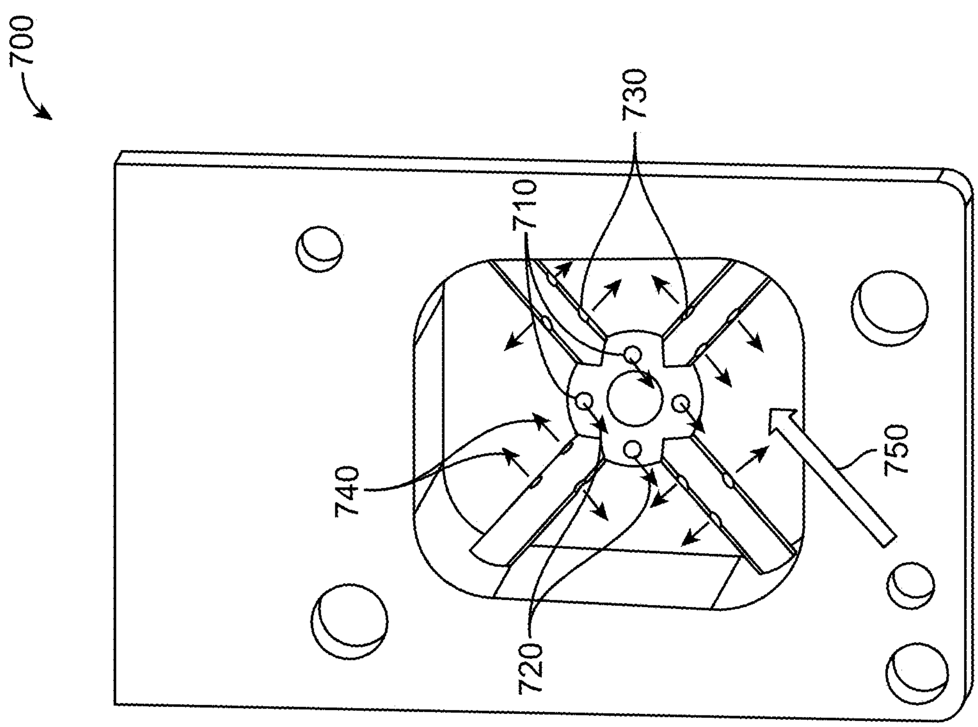

FIG. 7

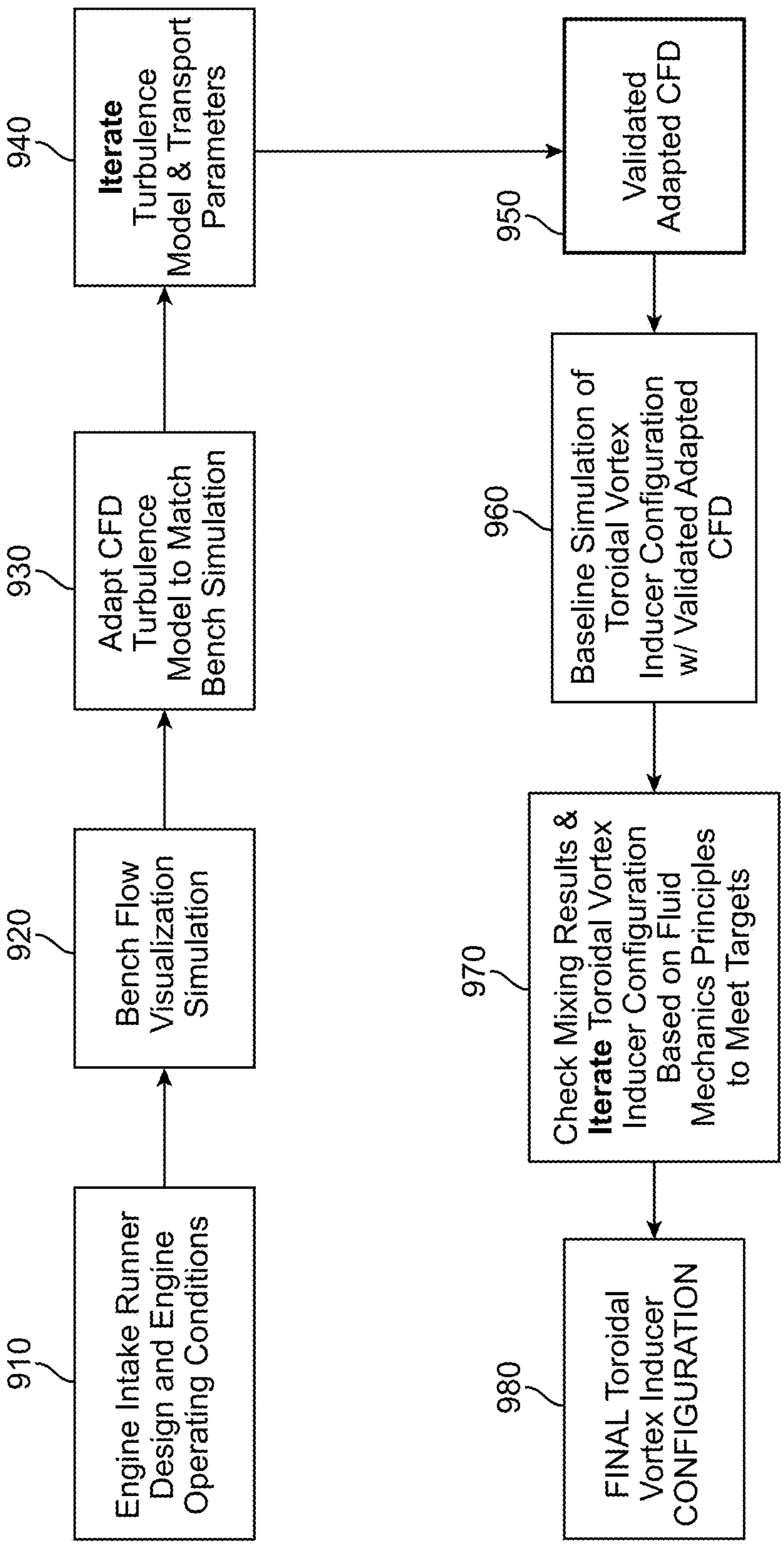
910
Engine Intake Runner Design and Engine Operating Conditions
920
Bench Flow Visualization Simulation
930
Adapt CFD Turbulence Model to Match Bench Simulation
940
Iterate Turbulence Model & Transport Parameters
950
Validated Adapted CFD
960
Baseline Simulation of Toroidal Vortex Inducer Configuration w/ Validated Adapted CFD
970
Check Mixing Results & Iterate Toroidal Vortex Inducer Configuration Based on Fluid Mechanics Principles to Meet Targets
980
FINAL Toroidal Vortex Inducer CONFIGURATION

FIG. 9

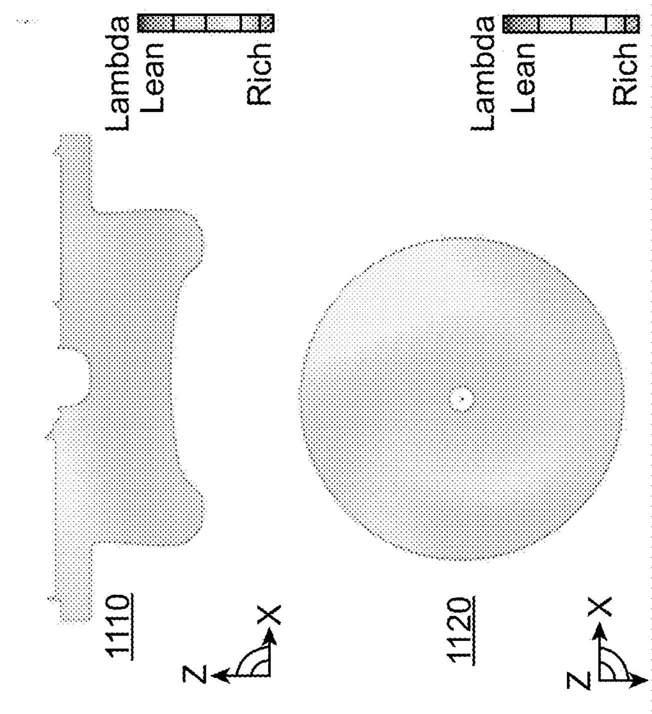
Lambda
Lean
Rich
1110
Z
X
Lambda
Lean
Rich
1120
Z
X
Y

FIG. 11

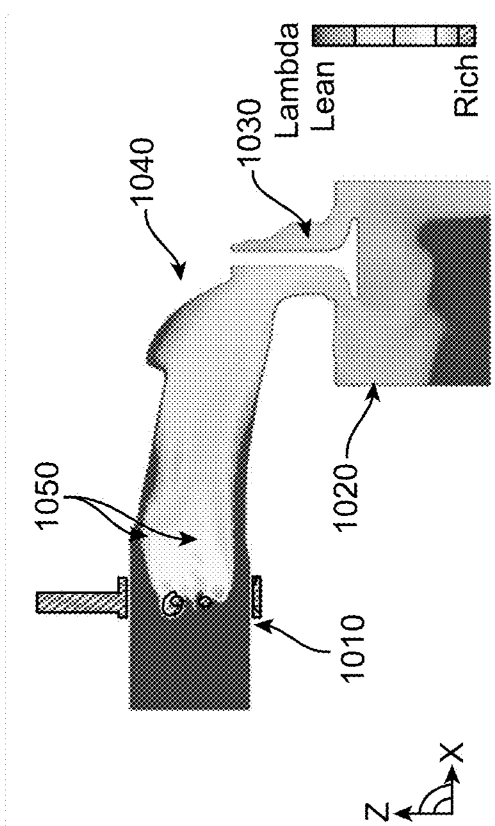
Lambda
Lean
Rich
1040
1030
1050
1020
1010
Z
X

FIG. 10

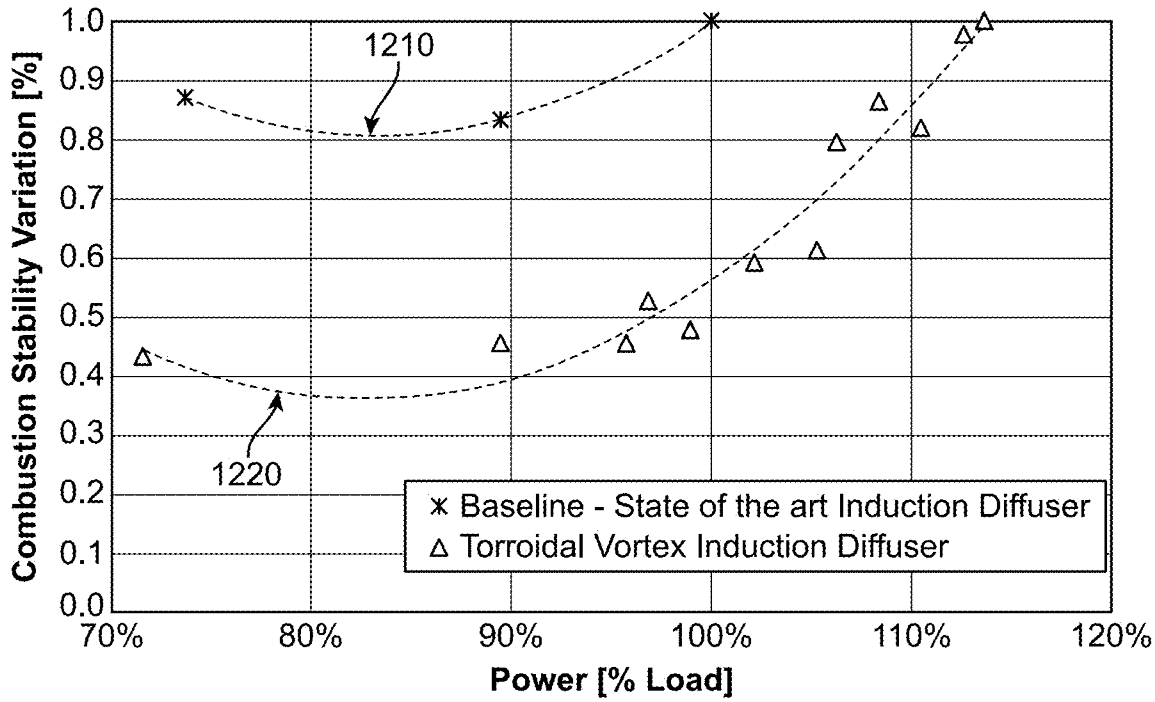
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
Combustion Stability Variation [%]
1210
1220
Baseline - State of the art Induction Diffuser
Torroidal Vortex Induction Diffuser
70%
80%
90%
100%
110%
120%
Power [% Load]

FIG. 12

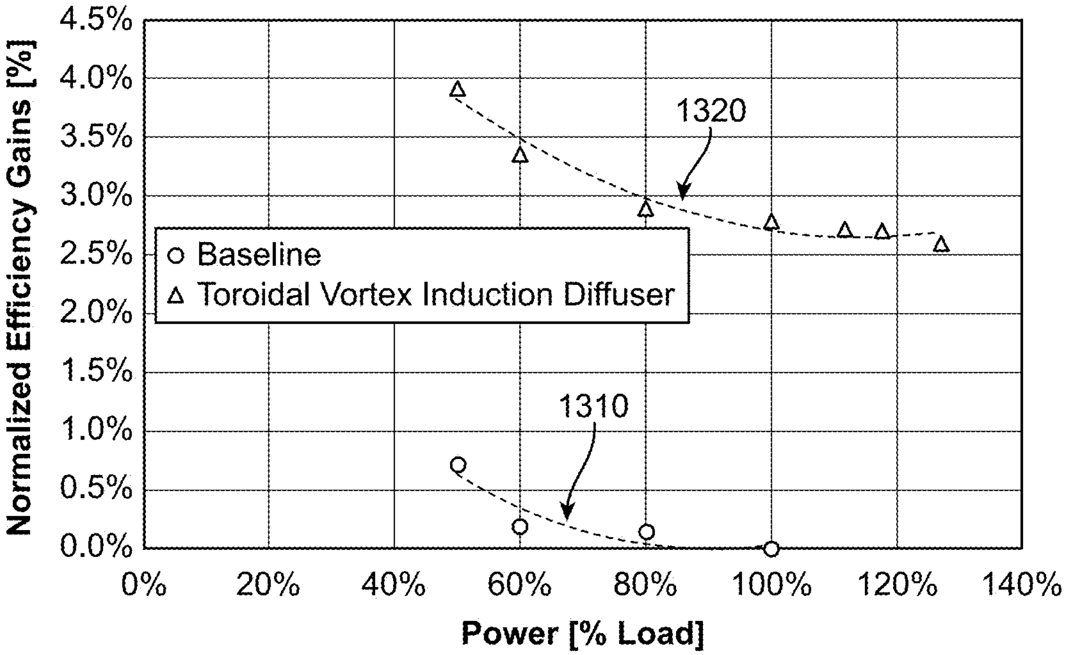
4.5%
4.0%
3.5%
3.0%
2.5%
2.0%
1.5%
1.0%
0.5%
0.0%
Normalized Efficiency Gains [%]
1320
1310
Baseline
Toroidal Vortex Induction Diffuser
0%
20%
40%
60%
80%
100%
120%
140%
Power [% Load]

FIG. 13

TOROIDAL VORTEX INDUCTION DIFFUSER

I. CROSS-REFERENCE TO RELATED APPLICATIONS

This application claims priority of U.S. Patent Application No. 63/706,490, entitled "Toroidal Vortex Induction Diffuser," and filed Oct. 11, 2024 and U.S. Patent Application No. 63/708,588, entitled "Toroidal Vortex Induction Diffuser," and filed Oct. 17, 2024. The entirety of the foregoing patent applications is incorporated by reference herein to the extent consistent with the present disclosure.

II. FIELD OF THE INVENTION

The disclosure generally relates to systems and methods for using toroidal vortex induction diffusers in internal combustion engines, and more particularly to methods and systems for using toroidal vortex induction diffusers to achieve high levels of fuel mixture homogeneity in combustion engines using hard-to-mix fuels like Hydrogen (H_2), Methanol (CH_3OH), Ethanol (C_2H_5OH) and other gaseous and liquid fuels.

III. BACKGROUND OF THE INVENTION

The following references describe problems with the state of the art that are more fully described below. These references are incorporated by reference herein to the extent consistent with this disclosure:

- [1] Sotiropoulou, E., et al: Enabling Diesel-Like Performance in Heavy Duty H_2-ICE via a Sophisticated Combustion System Solution, ATZ live Heavy Duty Engines 2024-MTZ Conference, Germany, 2024.
- [2] Sotiropoulou, E., et al: Same-Cycle Spark Control: The Future of Hydrogen Engines. 13th Dessau Gas Engine Conference, Dessau, Germany, 2024.
- [3] Sotiropoulou, E., et al: Holistic Solution Enabling High Power Density & Efficiency H_2-ICE, ATZ live Heavy Duty Engines 2023-MTZ Conference, Germany, 2023.
- [4] Sotiropoulou, E., et al: Prechamber Combustion: Enabling the Competitive Carbon-Neutral ICE. In CIMAC Congress, Paper No. 291, Busan, S. Korea, 2023.
- [5] Sotiropoulou, E., Tozzi, L., Trapp, C.: Breakthrough in Hydrogen Engine Combustion Enabling Zero Emissions and High Efficiency with Passive Prechamber Technology. In 12th Dessau Gas Engine Conference, Dessau, Germany 2022.
- [7] Yasueda, S., Sotiropoulou E., Tozzi, L.: Predicting Autoignition caused by Lubricating Oil in Gas Engines. In CIMAC Congress, Paper No. 37, Shanghai, China, 2013.
- [8] Sotiropoulou, et al: Prechamber Spark Plugs: The Evolution from Low Emission Natural Gas to Zero Emission H_2 Operation. MTZ Worldwide, vol. 2020-6, pp. 46-50, 2020.

Decarbonization and sustainability mandates pose serious uncertainties to the future of Internal Combustion Engines (ICEs).

Due to the challenges associated with combustion instabilities, the power density and efficiency of H_2 engines, also known as H_2 Internal Combustion Engines or H_2-ICEs, are much lower compared to that of Diesel engines.

Typical combustion issues with H_2 engines, defined as engines burning any fuel mixture combination that includes Natural Gas (NG), Ammonia (NH_3) and Hydrogen (H_2), or with carbon neutral fuels such as Methanol (CH_3OH), Ethanol (C_2H_5OH), synthetic gasoline, etc., are large variations in the Coefficient of Variation of Indicated Mean Effective Pressure (COV-IMEP), high propensity to Lubrication Oil Preignition (LOP), Knock and Preignition. Because of these limitations, only low levels of engine Indicated Mean Effective Pressure (IMEP) and Indicated Thermal Efficiency (ITE), and high levels of Nitrous Oxide (NOx) emissions can be obtained.

Current H_2-ICEs have performance parameters limited to the following ranges:

COV-IMEP>2%
IMEP≤16 bar
ITE≤41%
NOx≥100 mg/Nm3

However, the performance level needed for H_2-ICEs to compete with Hydrogen Fuel Cells would require performance parameters in the following ranges:

COV-IMEP≤1%
IMEP≥20 bar
ITE≥49%
NOx≤25 mg/Nm3

There is a need to address the foregoing deficiencies in the art.

IV. BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** depicts consecutive engine combustion cycles leading to backfire in the intake (frontfire) or preignition in accordance with certain embodiments.

FIG. **2** depicts heterogenous in-cylinder mixture air-fuel Lambda (λ) distribution with a conventional H_2 PFI system in accordance with certain embodiments.

FIG. **3** depicts homogeneous in-cylinder mixture **λ** distribution with advanced H_2 mixing system in accordance with certain embodiments.

FIG. **4** depicts an injection diffuser induction device having the fuel admission at the periphery of the intake runner in accordance with certain embodiments.

FIG. **5** depicts CFD simulation of heterogenous mixing in the intake port of an injection diffuser with fuel admission at the periphery of the intake runner in accordance with certain embodiments.

FIG. **6** depicts CFD simulation of heterogenous in-cylinder mixing of an injection diffuser with fuel admission at the periphery of the intake runner in accordance with certain embodiments.

FIG. **7** depicts an injection diffuser induction device having the fuel admission at the center of the intake runner or intake port in accordance with certain embodiments.

FIG. **8** depicts an injection diffuser induction device having the fuel admission at the center of the intake runner or intake port in accordance with certain embodiments.

FIG. **9** depicts a flow chart of the criteria and methodology to validate the CFD tool used to define the general configuration of the Toroidal Vortex Diffuser in relation to the engine intake runner and port geometries, the intake air flow direction, velocity and Reynolds number in accordance with certain embodiments.

FIG. **10** depicts a CFD simulation of toroidal vortex flow diffusing lean Lambda mixture through intake port and valve into the engine cylinder in accordance with certain embodiments.

FIG. **11** depicts CFD simulation results for the in-cylinder Lambda distribution resulting from the Toroidal Vortex Induction Diffuser in accordance with certain embodiments.

FIG. **12** depicts engine test measurements for a counter-flow Toroidal Vortex Induction Diffuser compared to a conventional induction diffuser in accordance with certain embodiments.

FIG. **13** depicts engine test measurements for a counter-flow Toroidal Vortex Induction Diffuser compared to a conventional induction diffuser in accordance with certain embodiments.

V. DETAILED DESCRIPTION

In certain embodiments, achieving levels of competitive efficiency and power density with zero-emissions, may necessitate holistic system solutions including the key components of the engine system like ignition, injection, and combustion chamber which may be properly integrated to meet or exceed performance targets. Certain embodiments may be able to burn green fuels like Hydrogen (H_2) while having engine power densities and efficiencies approaching those of Diesel engines (i.e., "Diesel-like performance") but with net-zero emissions.

In certain embodiments, validated CFD simulation tools may be used to design highly effective mixing devices to assure proper mixing during steady state, as well as during transient operations. In certain embodiments, the proper mixing of injected Hydrogen (H_2) and/or Methanol (CH_3OH) may be achieved.

In certain embodiments, performance levels may be achieved to enable H_2-ICEs to compete against Hydrogen Fuel Cells. In certain embodiments, the following performance parameters may be achieved:

COV-IMEP≤1%

IMEP≥20 bar

ITE≥49%

NOx≤25 mg/Nm3

In certain embodiments, highly homogeneous mixtures defined as mixtures having a Uniformity Index (UI) greater than 0.9 may be used to achieve the foregoing performance levels. Where the UI=1−(Standard Deviation of Lambda/Mean Lambda).

$$UI = 1 - \frac{\lambda_{st\,dev}}{\lambda_{mean}}$$

$UI = 1$, means 100% homogenous.

In certain embodiments, a toroidal induction diffuser for enhancing mixing of engine intake air and fuel in an engine cylinder is disclosed comprising: an engine intake port having a first longitudinal axis; an engine intake runner upstream of the engine intake port having a second longitudinal axis; a toroidal structure located around one of the first longitudinal axis and the second longitudinal axis; wherein the toroidal structure includes a first one or more holes for introducing a first fuel flow into the engine intake port or the engine intake runner. The first fuel flow may be in one of a direction aligned with, counter to, or transverse to a direction of flow of intake air to the engine intake runner or engine intake port.

The toroidal induction diffuser may further comprise one or more cross branch structures extending radially from the toroidal structure to one of the engine intake port or the engine intake runner, wherein the one or more cross branch structures may comprise a second one or more holes for introducing a second fuel flow into the engine intake port or the engine intake runner. The second fuel flow may be in one of a direction aligned with, counter to, or transverse to an intake air flow direction into the engine intake runner or engine intake port. The first fuel flow may be counter to the intake air flow direction and the second fuel flow may be transverse to the intake air flow direction. The first fuel flow may be aligned with the intake air flow direction and the second fuel flow may be transverse to the intake air flow direction. The first fuel flow may be counter to the intake air flow direction and the second fuel flow may be counter to the intake air flow direction. The first fuel flow may be aligned with the intake air flow direction and the second fuel flow may be aligned with the intake air flow direction. The first fuel flow may be counter to the intake air flow direction and the second fuel flow may be aligned with the intake air flow direction. The first fuel flow may be aligned with the intake air flow direction and the second fuel flow may be counter to the intake air flow direction.

The toroidal structure may be symmetric about at least one of the first longitudinal axis, the second longitudinal axis, or the intake air flow direction. The size of and geometry of the toroidal structure may be functions of one or more of engine intake runner and port geometries, intake air flow direction, velocity and Reynolds number. The size of and the geometry of the one or more cross branch structures may be functions one or more of engine intake runner and port geometries, intake air flow direction, velocity and Reynolds number. A size of, a geometry of, a number of and an orientation of the first one or holes may be functions of one or more of engine intake runner and port geometries, intake air flow direction, velocity and Reynolds number. A size of, a geometry of, a number of and an orientation of the second one or more holes may be functions of one or more of engine intake runner and port geometries, intake air flow direction, velocity and Reynolds number.

The fuel may comprise at least one of H_2, CH_3OH, and C_2H_5OH. A fuel-air mixture in the engine cylinder may have a Uniformity Index (UI) greater than about 0.9 according to the following equation:

$$UI = 1 - \frac{\lambda_{st\,dev}}{\lambda_{mean}}$$

The fuel-air mixture in the engine cylinder may have a Uniformity Index (UI) of about 0.98.

In certain embodiments, FIG. **1** shows a sequence of consecutive engine combustion cycles leading to backfire in the intake (i.e., frontfire). In certain embodiments, Cycle #**82** **110** shows a normal cycle. In certain embodiments Cycle #**83** **120** shows an advanced start of combustion (SOC) which may be caused by either hot spots from improper management of high energy ignition or by lube oil preignition (LOP). In certain embodiments, Cycle #**84** **130** shows an enhanced combustion Heat Released Rate cycle that may be caused by hot spots from the previous advanced SOC cycle **120**. In certain embodiments, Cycle #**85** **140** shows knock that may be caused by high temperatures from the preceding high combustion cycle **130**. In certain embodiments, Cycle #**86** **150** shows frontfire or preignition. This type of combustion instability may be typical in H_2 engines and may prevent reliable operation at higher power densities above 16 bar brake mean effective pressure (BMEP). In certain embodiments, the instability may be initiated by an advanced start of combustion (SOC) cycle, which may be caused by either hot spots from improper management of high energy ignition sparks or by the autoignition of lubricating oil droplets and may result in large cycle to cycle combustion variabilities.

In certain embodiments, the power density of an H_2 engine may be less than 60% of its Diesel counterpart due to these combustion instabilities. For example, if the base Diesel engine operates at 25 bar BMEP (brake mean effective pressure), the H_2 engine version may operate at less than 15 bar BMEP before the combustion becomes unstable. Accordingly, if for the Diesel version the engine efficiency is in the range of 48%, in the H_2 version the engine efficiency may be in the range of 40%.

As discussed in references [1]-[8] above, homogenous H_2 mixtures may prevent the occurrence of LOP at BMEP levels above 16 bar. Combustion CFD studies, as well as engine testing, have indicated that it is very difficult to thoroughly mix H_2 with air. This may be due to the incredibly small molecular mass of H_2 which limits the diffusion of H_2 in air, resulting in highly stratified mixing with conventional H_2 port fuel injection (PFI) systems, as shown in FIG. **2**. In certain embodiments, H_2-lean pockets **210** and H_2-rich pockets **220** may be created. In embodiments, H_2-rich pockets **220**, especially if located at the periphery of the cylinder where the concentration of lubricating oil tends to be higher, may result in LOP causing combustion instability in the form of abnormal high cylinder pressure cycles.

In contrast, certain embodiments shown in FIG. **3** may provide a more homogeneous in-cylinder H_2 mixture **310** Lambda (2=Actual Air-Fuel ratio/Stoichiometric Air-Fuel ratio), with leaner pockets in the periphery where lubricating oil is present. In these embodiments, the propagation of a flame front initiated by the local autoignition of the lubricating oil may be inhibited by the lean mixture which then may prevent combustion instabilities due, for example, to LOP. In certain embodiments, port fuel injection (PFI) injector diffuser induction devices may be used to achieve the necessary mixture homogeneity and, hence, to achieve the target power density and efficiency with H_2-ICEs.

In certain embodiments, a PFI diffuser induction device may use the principle of toroidal vortex flow to thoroughly mix H_2 and air in the intake runner and port of a H_2-ICE. Toroidal vortices in liquids and gases may enable thorough mixing in a two-component system, which in certain embodiments may include air and hydrogen. When H_2 enters a stream of air flow in the form of a toroidal vortex, it may tend to swallow the air into the vortex where a low-pressure region may be formed due to the swirling velocity of the vortex. This method of mixing may be more effective that the typical mixing via conventional injection.

In certain embodiments shown in FIG. **4**, an injection diffuser induction device with air intake direction **420**, may be characterized by having the fuel admission **410** at the periphery of the intake runner via injection through multiple holes **415**. In these embodiments, while the pressure drop across the diffuser may be minimized, the mixing may not be sufficiently homogeneous to achieve the required combustion stability target.

In certain embodiments, FIG. **5** shows a CFD simulation of the mixing in the intake port **510** of an H_2 engine showing that the mixture Lambda distribution **520** at the intake valve **530**, prior to entering the engine cylinder **540**, is not very uniform. In these embodiments, due to this inhomogeneity in the intake port **510**, the mixing in the cylinder may also be nonuniform as shown in piston side view **610** and cylinder head view from piston **620** of FIG. **6**, with a UI of approximately 0.9. In certain embodiments, this kind of inhomogeneity, having a UI~ **0.9**, may cause combustion instabilities like LOP which limit the engine operation to power density below 16 bar brake mean effective pressure (BMEP) as shown by extensive engine testing.

In certain embodiments as shown in FIGS. **7** and **8**, a toroidal injection diffuser induction device **700** may improve mixing by admitting fuel, for example H_2, from the center of the intake runner or intake port **830**, utilizing a toroidal structure **820** suspended about the center of the engine intake runner or port **830**. Unlike peripheral diffusers as shown in FIG. **4**, the fuel, for example H_2, may be induced from a central toroidal structure **820** having substantially radial cross branch structures **840** flow dynamically connected to the centrally located toroidal structure **820**. The centrally located toroidal structure **820** may have holes **710** delivering fuel **720** where the fuel flow may be concurrent, counter or transversal to the intake air flow **750** delivering the fuel streams in multiple directions. The radial cross branch structures **840** may have holes **730** delivering fuel **740** where the fuel flow may be concurrent, counter or transversal to the intake air flow **750** delivering the fuel streams in multiple directions. In certain embodiments, the direction of the fuel **720** released from the central toroidal structure **820** and the direction of the fuel **740** released from the cross branches **840** may be counter, concurrent, transversal or combinations of the said directions with respect to the direction of the intake air **750** stream. In the embodiment of FIGS. **7** and **8**, the directions of the fuel streams **740** delivered from the central toroidal structure **820** may be counter or concurrent to the air stream direction **750** and the direction of the fuel streams **740** delivered from the cross branches **840** may be transverse to the direction of the intake air stream **750**.

The arrangement shown in the embodiments of FIGS. **7** and **8** may enable the formation of a toroidal vortex flow in the midstream of the intake air flow **750** thereby enabling thorough mixing of hard-to-mix fuels. Hard-to-mix fuels may include Hydrogen (H_2, due to the very small molecular mass, or CH_3OH (Methanol), and C_2H_5OH (Ethanol), due to the relatively high viscosity and high latent heat of vaporization, and other gaseous and liquid fuels known to one of skill in the art. In certain embodiments, the lateral holes **730** located on the cross branches **840** and creating a fuel flow **740** that is approximately orthogonal to the direction of the intake air, may enable further mixing at the periphery of the intake channel.

In certain embodiments with reference to FIG. **8**, the general geometrical configuration of this injection diffuser induction device **700**, may be substantially symmetrical about the longitudinal axis **810** of the intake runner **830** or to the direction of the intake air stream **750** in order to promote uniform diffusion of the fuel in the intake air stream.

In certain embodiments with reference to FIG. **8**, a) the size and geometry of the central toroidal structure **820** and of the cross branches **840**, b) the size, geometry, number, orientation and location of the discharge orifices **710** on the central toroidal structure **820** and of the discharge orifices **730** on the cross branches **840**, and c) the number of the cross branches **840** may be functions of the engine intake runner and port **830** geometries, intake air flow direction **750**, velocity and Reynolds number (defined as Re=p u dh/u where p is the air density, u is the air velocity, the dh is the intake runner hydraulic diameter, and u is the air dynamic viscosity), injection pressure and duration and may be determined by means of a computational fluid dynamic (CFD) numerical tool that may be based on empirically validated physical models which may be representative of each engine type and application.

In certain embodiments, FIG. **9** is a flow chart of the process of validating a CFD tool that may be used to define the general configuration of the Toroidal Vortex Induction Diffuser in relation to the engine intake runner and port geometries, and intake air flow direction, velocity and Reynolds number.

In certain embodiments as shown in Step **910**, Engine Design and Operating Conditions may be input. In certain embodiments, a CFD turbulence model such as RNG Zeta-F or Reynolds Stress Model or Detached Eddies Simulation (DES) or Large Eddies Simulation (LES) may be adapted and the transport parameters for mass and heat such as Prandtl and Schmidt numbers may be adapted to the specific engine characteristics such as a) cylinder bore & stroke, b) engine rpm, c) intake runner geometry, d) Lambda, e) fuel type, and f) fuel injector geometry, to match the empirical data from bench flow visualization testing simulating engine operations as step **930** for each application segment such as industrial, marine and heavy duty on-off highway.

In certain embodiments, the adapted CFD turbulence model may be used to simulate the intake and compression processes in terms of flow and Lambda distribution. In certain embodiments as shown in Step **920**, a first simulation of the current engine configuration may be conducted to establish a baseline. In certain embodiments, the target improvements in terms of mixture homogeneity and required mixing levels may be formulated based on results of the simulation. In certain embodiments, a strategy may be defined to achieve the targeted improvements in terms of type and geometry of the injector, injection pressure and duration, and Toroidal Vortex Induction Diffuser design and location in the intake runner or port. In certain embodiments as shown in Step **940**, iterative simulations of the configuration resulting from the initial strategy may be conducted and the results may be analyzed in terms of mixture homogeneity to arrive at validated adapted CFD **950**. In certain embodiments as shown in step **960**, a baseline simulation of the toroidal vortex inducer configuration may be conducted using validated adapted CFD **950**. Certain embodiments as shown in step **970** may further improve the system in terms of Toroidal Vortex Induction Diffuser design and location in the intake runner or port and injector design and location in the intake runner or port as needed based on fundamental fluid mechanics principles based on results of the iterative simulations to arrive at final toroidal vortex inducer configuration **980**.

In certain embodiments, FIG. **10** shows validated CFD simulation results for a toroidal vortex diffuser. In certain embodiments during the intake process, the toroidal vortex flow **1050** located at the center of the intake runner **1010** may diffuse lean mixture Lambda through intake port **1040** and valve **1030** into engine cylinder **1020**.

In certain embodiments, FIG. **11** shows a piston side view **1110** and a cylinder head view **1120** from piston of CFD simulation results for the in-cylinder Lambda distribution resulting from a toroidal vortex induction diffuser.

In certain embodiments with the Toroidal Vortex Induction Diffuser, the in-cylinder mixture Lambda distribution may be nearly homogeneous with a uniformity index UI=0.98, which is a substantial improvement compared to the peripheral induction diffuser portrayed in FIGS. **4**, **5**, and **6** and having a UI=0.9.

In certain embodiments, FIG. **12** shows engine measurement data describing the relationship between the combustion stability variation [%], in terms of LOP intensity and frequency of occurrence, calculated as coefficient of variation of indicated mean effective pressure (COVIMEP [%]), and engine power output [KW] for engine test results of the Toroidal Vortex Induction Diffuser **1220** compared to a conventional Induction Diffuser **1210** As shown in FIG. **12**, the Toroidal Vortex Induction Diffuser **1220** combustion stability may be substantially increased resulting in a power output improvement of approximately 14% measured at 1% combustion stability variation.

In certain embodiments, FIG. **13** shows engine measurement data describing the relationship between efficiency, calculated as Normalized Efficiency Gains [%], and Power [% Load] for engine test results of the Toroidal Vortex Induction Diffuser **1320** compared to a conventional Induction Diffuser **1310**. As shown in FIG. **13**, the Toroidal Vortex Induction Diffuser **1320** may enable better than 3% engine efficiency, measured in brake thermal efficiency (BTE), and more than 25% in power output. In certain embodiments, engine test measurements showing remarkable improvements in engine efficiency and power output enabled by a novel counterflow Toroidal Vortex Induction Diffuser achieving high levels of fuel mixture homogeneity in combustion engines using hard to mix fuels like H_2, CH_3OH, C_2H_5OH and other gaseous and liquid fuels.

In certain embodiments, a conventional Induction Diffuser may admit fuel laminarly at the periphery of the intake channel resulting in poor mixing, whereas a Toroidal Vortex Induction Diffuser may admit fuel in the center of the intake runner with a toroidal vortex flow which enables thorough mixing of the injected fuel with the intake air. In certain embodiments, the toroidal vortex flow may result in a highly homogeneous mixture being created inside the cylinder which may enable improvement in engine combustion stability, power output, engine efficiency and transient response. In certain embodiments, improvements in engine efficiency (BTE) and transient response to load changes are expected to follow the trend of the improvements measured in power output.

While the invention has been described with reference to the specific embodiments thereof, it should be understood by those skilled in the art that various changes may be made, and equivalents may be substituted without departing from the true spirit and scope of the invention as defined by the appended claims. In addition, many modifications may be made to adapt a particular situation, material, composition of matter, method, operation or operations, to the objective, spirit, and scope of the invention. All such modifications are intended to be within the scope of the claims appended hereto. In particular, while the methods disclosed herein have been described with reference to particular operations performed in a particular order, it will be understood that these operations may be combined, sub-divided, or reordered to form an equivalent method without departing from the teachings of the invention. Accordingly, unless specifically indicated herein, the order and grouping of the operations are not a limitation of the invention.

We claim:

1. A toroidal induction diffuser for enhancing mixing of engine intake air and fuel in an engine cylinder, comprising:

an engine intake port having a first longitudinal axis;

an engine intake runner upstream of the engine intake port having a second longitudinal axis;

a toroidal structure located around one of the first longitudinal axis and the second longitudinal axis;

wherein the toroidal structure includes a first one or more holes for introducing a first fuel flow into the engine intake port or the engine intake runner; and

one or more cross branch structures extending radially from the toroidal structure to one of the engine intake port or the engine intake runner, wherein the one or more cross branch structures comprise a second one or more holes for introducing a second fuel flow into the engine intake port or the engine intake runner.

2. The toroidal induction diffuser of claim **1**, wherein the first fuel flow is in one of a direction aligned with, counter to, or transverse to a direction of flow of intake air to the engine intake runner or engine intake port.

3. The toroidal induction diffuser of claim **1**, wherein the second fuel flow is in one of a direction aligned with, counter to, or transverse to an intake air flow direction into the engine intake runner or engine intake port.

4. The toroidal induction diffuser of claim **1**, wherein the first fuel flow is counter to the intake air flow direction and the second fuel flow is transverse to the intake air flow direction.

5. The toroidal induction diffuser of claim **1**, wherein the first fuel flow is aligned with the intake air flow direction and the second fuel flow is transverse to the intake air flow direction.

6. The toroidal induction diffuser of claim **1**, wherein the first fuel flow is counter to the intake air flow direction and the second fuel flow is counter to the intake air flow direction.

7. The toroidal induction diffuser of claim **1**, wherein the first fuel flow is aligned with the intake air flow direction and the second fuel flow is aligned with the intake air flow direction.

8. The toroidal induction diffuser of claim **1**, wherein the first fuel flow is counter to the intake air flow direction and the second fuel flow is aligned with the intake air flow direction.

9. The toroidal induction diffuser of claim **1**, wherein the first fuel flow is aligned with the intake air flow direction and the second fuel flow is counter to the intake air flow direction.

10. The toroidal induction diffuser of claim **1**, wherein the toroidal structure is symmetric about the first longitudinal axis, the second longitudinal axis, or the intake air flow direction.

11. The toroidal induction diffuser of claim **1**, wherein the size of and geometry of the toroidal structure are functions of one or more of engine intake runner and port geometries, intake air flow direction, velocity and Reynolds number.

12. The toroidal induction diffuser of claim **1**, wherein a size of and a geometry of the one or more cross branch structures are functions of one or more of engine intake runner and port geometries, intake air flow direction, velocity and Reynolds number.

13. The toroidal induction diffuser of claim **1**, wherein a size of, a geometry of, a number of and an orientation of the first one or holes are functions of one or more of engine intake runner and port geometries, intake air flow direction, velocity and Reynolds number.

14. The toroidal induction diffuser of claim **1**, wherein a size of, a geometry of, a number of and an orientation of the second one or more holes are functions of one or more of engine intake runner and port geometries, intake air flow direction, velocity and Reynolds number.

15. The toroidal induction diffuser of claim **1**, wherein the fuel comprises at least one of H_2, CH_3OH, and C_2H_5OH.

16. The toroidal induction diffuser of claim **1**, wherein a fuel-air mixture in the engine cylinder has a Uniformity Index (UI) greater than about 0.9 according to the following equation:

$$UI = 1 - \frac{\lambda_{st\,dev}}{\lambda_{mean}}.$$

17. The toroidal induction diffuser of claim **16**, wherein the fuel-air mixture in the engine cylinder has a Uniformity Index (UI) of about 0.98.

18. The toroidal induction diffuser of claim **1**, wherein a fuel-air mixture in the engine cylinder has a Uniformity Index (UI) greater than about 0.9 according to the following equation:

$$UI = 1 - \frac{\lambda_{st\,dev}}{\lambda_{mean}}.$$

19. The toroidal induction diffuser of claim **18**, wherein the fuel-air mixture in the engine cylinder has a Uniformity Index (UI) of about 0.98.

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